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Buckling behavior of rectangular plates under uniaxial and biaxial compression

Mohamed Bourada[1], Abed Bouadi[2,3], Abdelmoumen Anis Bousahla[2,3], Amel Senouci[1], Fouad Bourada*[1,4], Abdelouahed Tounsi[1,5] and S.R. Mahmoud[6]

[1]*Material and Hydrology Laboratory, University of Sidi Bel Abbes, Faculty of Technology, Civil Engineering Department, Algeria*
[2]*Laboratoire de Modélisation et Simulation Multi-échelle, Département de Physique, Faculté des Sciences Exactes, Département de Physique, Université de Sidi Bel Abbés, Algeria*
[3]*Centre Universitaire Ahmed Zabana de Relizane, Algeria*
[4]*Département des Sciences et de la Technologie, centre universitaire de Tissemsilt, BP 38004 Ben Hamouda, Algérie*
[5]*Department of Civil and Environmental Engineering, King Fahd University of Petroleum & Minerals, 31261 Dhahran, Eastern Province, Saudi Arabia*
[6]*Department of Mathematics, Faculty of Science, King Abdulaziz University, Jeddah, Saudi Arabia*

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Abstract. In the classical stability investigation of rectangular plates the classical thin plate theory (CPT) is often employed, so omitting the transverse shear deformation effect. It seems quite clear that this procedure is not totally appropriate for the investigation of moderately thick plates, so that in the following the first shear deformation theory proposed by Meksi *et al.* (2015), that permits to consider the transverse shear deformation influences, is used for the stability investigation of simply supported isotropic rectangular plates subjected to uni-axial and bi-axial compression loading. The obtained results are compared with those of CPT and, for rectangular plates under uniaxial compression, a novel direct formula, similar to the conventional Bryan's expression, is found for the Euler stability stress. The accuracy of the present model is also ascertained by comparing it, with model proposed by Piscopo (2010).

Keywords: buckling stress; isotropic plate; FSDT; Navier method

1. Introduction

The buckling of the rectangular plate is phenomenon of instability which occurs when the plate is subjected to an axial compression load. Most commonly applied loads are uni-axial and bi-axial loads. Several studies on the buckling analysis of the plate/beam were published by Bank and Jin (1996), Kang and Leissa (2005), Hwang and Lee (2006), Matsunaga (2009), Kim *et al.* (2009), Bourada *et al.* (2012), Altunsaray and Bayer (2014), Meziane *et al.* (2014), Afsharmanesh *et al.* (2014), Swaminathan and Naveenkumar (2014), Panda and Katariya (2015), Nguyen *et al.*(2015), Bouguenina *et al.* (2015), Tebboune *et al.* (2015), Rajanna *et al.* (2016), Yousefitabar and Matapouri (2017), Houari *et al.* (2016). Musa (2016), Katariya and Panda (2016), Arani and Kolahchi (2016), Eltaher *et al.* (2016), Bourada *et al.* (2016), Bouderba *et al.* (2016), Kolahchi and Bidgoli (2016), Bousahla *et al.* (2016), Kolahchi *et al.* (2016ab), Bilouei *et al.* (2016), Kolahchi *et al.* (2017ab), Hajmohammad*et al.* (2017), Abdelaziz *et al.* (2017), Sekkal *et al.* (2017a), El-Haina *et al.* (2017), Zamanian *et al.* (2017), Kolahchi and Cheraghbak (2017), Kolahchi (2017), Yazid *et al.* (2018), Fakhar and Kolahchi (2018), Kadari *et al.* (2018), Bourada *et al.* (2018), Golabchi *et al.* (2018), Mokhtar *et al.* (2018), Shahsavari *et al.* (2018), Shahsavari *et al.* (2018a, b), Karami *et al.* (2018a, b, c). Generally, the problem of stability of thin plates is solved by applying the Love-Kirshoff model (classical plate theory) which neglects the transverse shear effect. For buckling analysis of thick and moderately thick plate, the classical plate theory (CPT) will no longer be retained. Hence, a new formulation which describes the shear effect through the thickness was required. For this purpose Reissner (1945) and Mindlin (1951) proposed a first shear deformation plate theory (FSDT), taking into account the transverse shear effect with uniform distribution through the thickness of the plate. Several works has been carried out on the buckling study of the plate based on the FSDT, such as Lanhe (2004), Yaghoobi and Torabi (2013), Mohammadi *et al.* (2010), Bouazza *et al.* (2010), Zarei *et al.* (2017), Madani *et al.* (2017), Amnieh *et al.* (2018), Youcef *et al.* (2018), Karami *et al.* (2018a) and Hajmohammad *et al.* (2018a). Recently, a new first shear deformation plate theory based on the model proposed by Shimpi (2002) was developed by Meksi *et al.* (2015) and Bellifa *et al.* (2016).

In this paper, the buckling analysis of isotropic rectangular plate under the action of uni-axial and bi-axial compressive stresses using the model of Meksi *et al.* (2015). In this model the transverse shear stresses is

*Corresponding author, Ph.D.
E-mail: bouradafouad@yahoo.fr

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considered to be uniform through the thickness of the plate, which necessitates the introduction of a shear correction factor. The theory of Meksi *et al.* (2015) has been modified in order to have only two variables and two equilibrium equations. The equilibrium equations are determined using the principle of virtual works. The results obtained are compared with those presented by Piscopo (2010) based on the model of Shimpi (2002) and the Bryan's expression for simply supported thin rectangular plates.

2. Theoretical formulation

The Bryan formula for the Euler buckling stress can be written in the form (Piscopo 2010)

$$\sigma_E = \frac{\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2\left(\frac{\alpha}{m}+\frac{m}{\alpha}\right)^2 \tag{1}$$

E, ν, *m* and α are Young modulus, Poisson coefficient, the number of half-waves in the direction of compression and dimension ratio ($\alpha=a/b$) respectively. The magnitude of the Euler load depends on the dimension ratio α, mode of the plate *m* in to which the plate buckles. The Bryan formula Eq. (1) can be rewritten as follow (Piscopo 2010)

$$\sigma_E = \frac{\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2 K_1 \tag{2}$$

Where K_1 is buckling factor defined as (Piscopo 2010)

$$K_1 = \begin{cases} 4.00 & if \quad \alpha > 1 \\ \left(\alpha + \dfrac{1}{\alpha}\right)^2 & if \quad \alpha \le 1 \end{cases} \tag{3}$$

In the following, a new formula of the Euler buckling stress which differs from the Bryan's formula by introduction of a corrective function will be presented. Finally, some applications are presented for the isotropic rectangular plate subjected to axial compression load along the *x* and *y* directions.

2.1 Basic assumptions of FSDT

-The displacements are small in comparison with the plate thickness t.

-The displacement *u* in *x* direction and *v* in *y* direction consists of extension and bending components

$$u = u_0 + u_b \tag{4a}$$

$$v = v_0 + v_b \tag{4b}$$

-The bending component u_b and v_b are similar to those of CPT (Karami *et al.* 2018b, Shahsavari *et al.* 2017, 2018) and can be given as

$$u_b = -z\frac{\partial w_b}{\partial x}; v_b = -z\frac{\partial w_b}{\partial y} \tag{5}$$

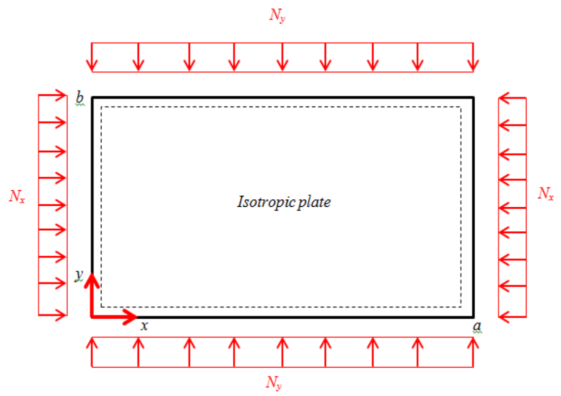

Fig. 1 Isotropic rectangular plate subjected to in-plane loading

-The transverse normal stress σ_z is negligible in comparison with in plane stresses σ_x and σ_y.

-The vertical displacement w includes two components of bending w_b and shear w_s.

$$w(x,y) = w_b(x,y) + w_s(x,y) \tag{6}$$

2.2 Kinematics

Based on the assumptions made in the preceding section, the displacements field can be obtained using Eqs. (4)-(6)

$$u(x,y,z) = u_0(x,y) - z\frac{\partial w_b}{\partial x} \tag{7a}$$

$$v(x,y,z) = v_0(x,y) - z\frac{\partial w_b}{\partial y} \tag{7b}$$

$$w(x,y) = w_b(x,y) + w_s(x,y) \tag{7c}$$

The principle of virtual works is used here to derive the equilibrium equations, the principle can be declared in the analytical form (Bouderba *et al.* 2013, Tounsi *et al.* 2013, Zidi *et al.* 2014, Zemri *et al.* 2015, Boukhari *et al.* 2016, Bounouara *et al.* 2016, Besseghier *et al.* 2017, Zidi *et al.* 2017, Chikh *et al.* 2017, Mouffoki *et al.* 2017, Khetir *et al.* 2017, Klouche *et al.* 2017, Fahsi*et al.* 2017, Bourada *et al.* 2019)

$$\int_V (\delta U + \delta V)dz = 0 \tag{8}$$

Where δU, δV, the variation of the strain energy and variation of works of the externals forces. The governing equations can be obtained in the following form

$$D\nabla^4 w_b = N_x\frac{\partial^2 w}{\partial x^2} + N_y\frac{\partial^2 w}{\partial y^2} + N_{xy}\frac{\partial^2 w}{\partial x \partial y} \tag{9a}$$

$$k_s Gt\left(\frac{\partial^2 w_s}{dx^2}+\frac{\partial^2 w_s}{dy^2}\right)=N_x\frac{\partial^2 w}{\partial x^2}+N_y\frac{\partial^2 w}{\partial y^2}+N_{xy}\frac{\partial^2 w}{\partial x\partial y} \quad (9b)$$

Where

$$D=\frac{Et^3}{12(1-\nu^2)};G=\frac{E}{2(1+\nu)} \quad (10)$$

N_x, N_y, N_{xy} are in-plane distributed forces. Assuming that those forces are constant along the plate, with N_{xy}=0, N_y= γN_x, such as 0≤λ≤1. Eq. (9) can be written as follows

$$D\nabla^4 w_b = N_x\left(\frac{\partial^2 w}{\partial x^2}+\gamma\frac{\partial^2 w}{\partial y^2}\right) \quad (11a)$$

$$k_s Gt\left(\frac{\partial^2 w_s}{dx^2}+\frac{\partial^2 w_s}{dy^2}\right)=N_x\left(\frac{\partial^2 w}{\partial x^2}+\gamma\frac{\partial^2 w}{\partial y^2}\right) \quad (11b)$$

The Navier method is only applied for simply supported plate on all four edges, as shown in Fig. 1.

The following displacements functions w_b and w_s are chosen to automatically satisfy the boundary conditions.

$$w_b=\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}W_{bmn}\sin(\lambda x)\sin(\mu y) \quad (12a)$$

$$w_s=\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}W_{smn}\sin(\lambda x)\sin(\mu y) \quad (12b)$$

Where

$$\lambda=\frac{m\pi}{a},\mu=\frac{n\pi}{b} \quad (13)$$

W_{bmn}, W_{smn} are arbitrary parameters to be determined.

Substituting Eq. (12) into Eq. (11), the following system is obtained as

$$A_{11}W_{bmn}+A_{12}W_{smn}=0 \quad (14a)$$

$$A_{21}W_{bmn}+A_{22}W_{smn}=0 \quad (14b)$$

With

$$\begin{aligned}
A_{11}&=D\pi^2\left(\frac{m^2}{a^2}+\frac{n^2}{b^2}\right)^2-N_x\left(\frac{m^2}{a^2}+\gamma\frac{n^2}{b^2}\right)\\
A_{12}&=-N_x\left(\frac{m^2}{a^2}+\gamma\frac{n^2}{b^2}\right)\\
A_{21}&=A_{12}\\
A_{22}&=\frac{k_s Et}{2(1+\nu)}-N_x\left(\frac{m^2}{a^2}+\gamma\frac{n^2}{b^2}\right)
\end{aligned} \quad (14c)$$

The Euler buckling stress can be determined by solve the system |A|t=0, such as

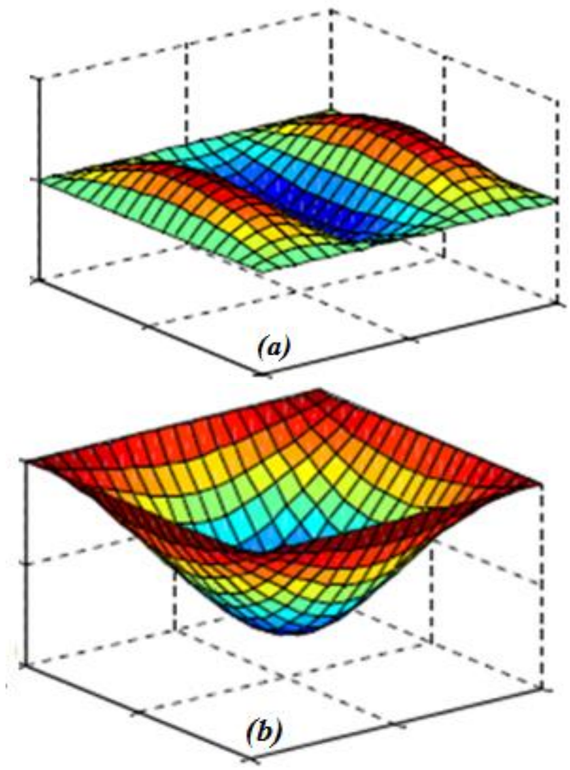

Fig. 2 Buckling modes shapes of isotropic rectangular plate: (a) several half wave (m≥1) in the direction of compression and one half wave in the perpendicular direction (n=1), (b) one half wave in the both direction (m=1, n=1)

$$A=\begin{bmatrix}A_{11} & A_{12}\\ A_{21} & A_{22}\end{bmatrix} \quad (15)$$

This makes it possible to obtain

$$\sigma_E=\frac{\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2\frac{\left(\frac{m^2}{\alpha^2}+n^2\right)^2}{\frac{m^2}{\alpha^2}+\gamma n^2}F(k_s,\frac{t}{b},m,n,\nu,\alpha) \quad (16a)$$

With

$$F(k_s,\frac{t}{b},m,n,\nu,\alpha)=\frac{6k_s b^2(1-\nu)}{(t\pi)^2\left(\frac{m^2}{\alpha^2}+n^2\right)+6k_s b^2(1-\nu)} \quad (16b)$$

k_s is shear correction factor. The Euler buckling stress σ_E, for isotropic plates under uni-axial compression loading (γ=0), can be derived considering the plate buckles in such a way that can be several half wave (m≥1) in the direction of compression, but only one half wave in the perpendicular direction (n=1), as shown Fig. 2.

A new expression of the Euler buckling stress, similar to the Bryan formula and differing by the corrective function $F(k_s, t/b, m, n, \nu, \alpha)$ is obtained

$$\sigma_E=\frac{\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2\left(\frac{\alpha}{m}+\frac{m}{\alpha}\right)^2F(k_s,\frac{t}{b},m,n,\nu,\alpha) \quad (17a)$$

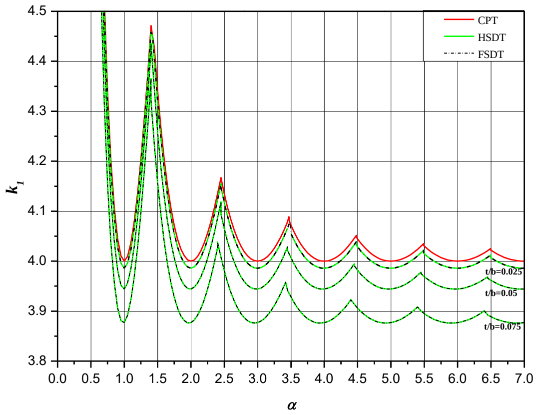

Fig. 3 Buckling factor k_1 for thick-plates under uni-axial compression (ν=0.30; γ=0)

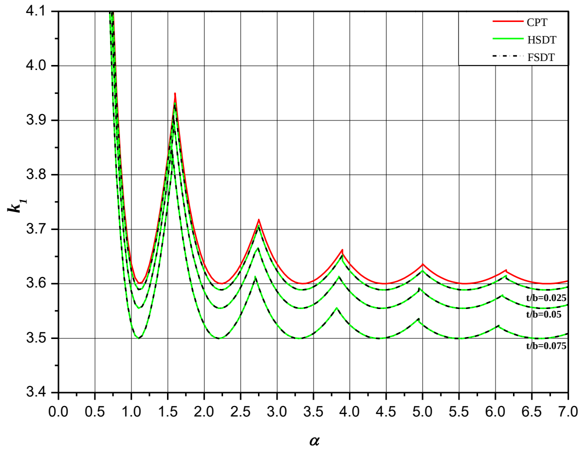

Fig. 4 Buckling factor k_1 for thick-plates under biaxial compression (ν=0.30; γ=0.1)

With

$$F(k_s,\frac{t}{b},m,n,\nu,\alpha)=\frac{6k_s b^2(1-\nu)}{(t\pi)^2\left(\dfrac{m^2}{\alpha^2}+1\right)+6k_s b^2(1-\nu)} \tag{17b}$$

Clearly, when the effect of transverse shear deformation is neglected. The corrective function becomes equal to one $F(k_s, t/b, m, n, \nu, \alpha)$=1, which gives the classical Bryan's formula for thin plates.

3. Results and discussions

In the following, the buckling factor of isotropic rectangular plates under uni-axial compression load defined in the Eq. (2) will be presented in the form of explicated graphs. In Fig. 3, several curves of the buckling factor as a function of the geometry ratio (t/b) and dimension ratio α are presented. It can be seen that a good agreement between the results obtained by the present formula based on the first shear order deformation plate theory with k_s=5/6 (Mindlin 1951, Menaa *et al.* 2012, Bouderba *et al.* 2016) and those obtained by the formula of Piscopo (2010) based on high order shear deformation plate theory.

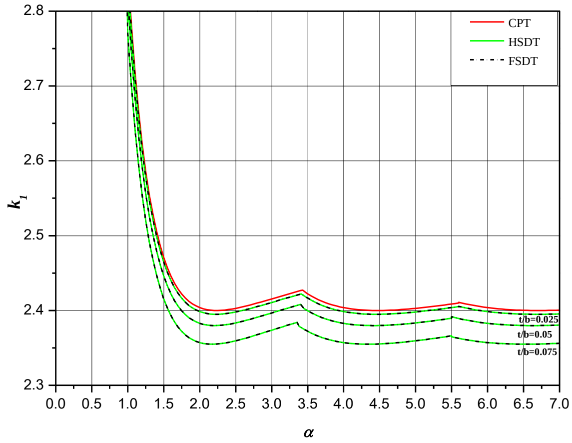

Fig. 5 Buckling factor k_1 for thick-plates under biaxial compression (ν=0.30; γ=0.4)

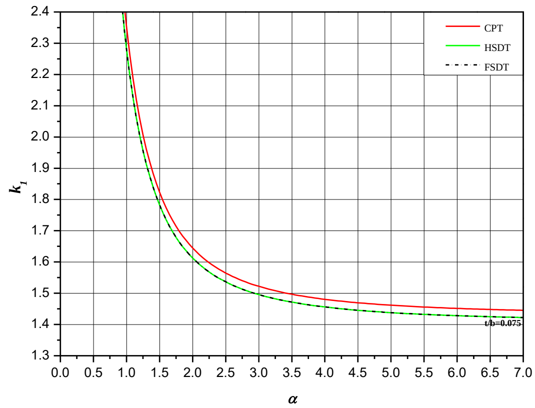

Fig. 6 Buckling factor k_1 for thick-plates under biaxial compression (ν=0.30; γ=0.7)

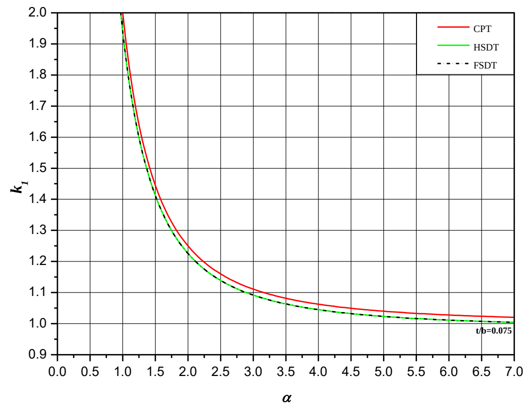

Fig. 7 Buckling factor k_1 for thick-plates under biaxial compression (ν=0.30; γ=1.0)

It should be noted that for plate under biaxial compressive loads with $0<N_y<0.5N_x$ in Figs. 4 and 5, the plate buckle in such a way that several half waves in the direction of the application of the greatest compressive forces and this number of half waves increase with increasing of the dimension ratio α, whereas a single half wave in the direction perpendicular to direction of applications of the largest compressive forces.

Table 1 The Euler buckling load N_E of isotropic rectangular plate ($Cas1-\alpha=0.25$, $\gamma=0$)

t/b	Mean element length	ANSYS Piscopo (2010) (A)	CPT Piscopo (2010) (B)	HSDT Piscopo (2010) (C)	FSDT Present (D)	$\frac{B-A}{A}*100$	$\frac{C-A}{A}*100$	$\frac{D-A}{A}*100$
---	m	KN/m	KN/m	KN/m	KN/m	%	%	%
0.01	0.050	3553	3363	3347	3347	0.87	0.39	0.39
	0.025	3391						
	0.010	3343						
	0.005	3334						
0.02	0.050	27924	26904	26398	26398	2.80	0.87	0.87
	0.025	26623						
	0.010	26233						
	0.005	26171						
0.03	0.050	91643	90800	87046	87045	5.66	1.30	1.30
	0.025	87372						
	0.010	86121						
	0.005	85932						

Table 2 The Euler buckling load N_E of isotropic rectangular plate ($Cas2-\alpha=1.00$, $\gamma=0$)

t/b	Mean element length	ANSYS Piscopo (2010) (A)	CPT Piscopo (2010) (B)	HSDT Piscopo (2010) (C)	FSDT Present (D)	$\frac{B-A}{A}*100$	$\frac{C-A}{A}*100$	$\frac{D-A}{A}*100$
---	m	KN/m	KN/m	KN/m	KN/m	%	%	%
0.01	0.050	747	745	744	744	0.68	0.54	0.54
	0.025	743						
	0.010	741						
	0.005	740						
0.02	0.050	5930	5958	5945	5945	1.95	1.73	1.73
	0.025	5883						
	0.010	5852						
	0.005	5844						
0.03	0.050	19811	20108	20006	20006	3.14	2.62	2.62
	0.025	19616						
	0.010	19513						
	0.005	19496						

Rather, when $0.5N_x<N_y<N_x$ the plate buckle in such a way as a single half wave in both directions x and y (Figs. 6 and 7). For scaling reasons only the curves of the thick plates are presented. It can be noted that the results of the present model are almost in line with those obtained by Piscopo (2010).

3.1 Numerical applications

In this part, the Euler buckling load ($N_E=\sigma_E t$) of isotropic rectangular plates subjected to uni-axial and bi-axial compression forces is presented to show the feasibility of the present formula. A comparison is made between the results obtained by the present model and those obtained by Piscopo (2010) using the high order shear deformation plate theory and the finite element method carried out by software ANSYS.

The Tables 1 and 2 below show the Euler buckling force per unit of length of an isotropic plate ($E=2.06\times10^{11}$, $v=0.3$, $b=1m$) under uni-axial compression loading along the x axis as a function of the geometry ratio (t/b). The results of the present formula with $k_s=5/6$ are in good agreement with those obtained by Piscopo's formula and the finite element method. It should be noted that the percentage difference of theoretical results increases with the increase of the thickness of the plate.

In Tables 3, 4 and 5, the Euler buckling force N_E per unit of length of rectangular isotropic plates under bi-axial

Table 3 The Euler buckling load N_E of isotropic rectangular plate ($Cas1-\alpha=0.25$, $\gamma=1.0$)

t/b	Mean element length	ANSYS Piscopo (2010) (A)	CPT Piscopo (2010) (B)	HSDT Piscopo (2010) (C)	FSDT Present (D)	$\frac{B-A}{A}*100$	$\frac{C-A}{A}*100$	$\frac{D-A}{A}*100$
---	m	KN/m	KN/m	KN/m	KN/m	%	%	%
0.01	0.050	3356	3165	3150	3150	0.86	0.38	0.38
	0.025	3194						
	0.010	3147						
	0.005	3138						
0.02	0.050	26377	25321	24845	24845	2.80	0.87	0.87
	0.025	25080						
	0.010	24693						
	0.005	24631						
0.03	0.050	86563	85459	81926	81924	5.67	1.30	1.30
	0.025	82304						
	0.010	81063						
	0.005	80875						

Table 4 The Euler buckling load N_E of isotropic rectangular plate ($Cas2-\alpha=1.00$, $\gamma=1.0$)

t/b	Mean element length	ANSYS Piscopo (2010) (A)	CPT Piscopo (2010) (B)	HSDT Piscopo (2010) (C)	FSDT Present (D)	$\frac{B-A}{A}*100$	$\frac{C-A}{A}*100$	$\frac{D-A}{A}*100$
---	m	KN/m	KN/m	KN/m	KN/m	%	%	%
0.01	0.050	373	372	372	372	0.54	0.54	0.54
	0.025	371						
	0.010	370						
	0.005	370						
0.02	0.050	2965	2979	2972	2972	1.95	1.71	1.71
	0.025	2942						
	0.010	2926						
	0.005	2922						
0.03	0.050	9906	10054	10003	10003	3.14	2.62	2.62
	0.025	9809						
	0.010	9758						
	0.005	9748						

compressive loads (γ=1) for the different values of the geometry ratio (t/b) are presented.

It can be seen that the classical plate theory provides the larger values of the Euler buckling load N_E to those obtained by FSDT and HSDT theories. This is due to neglect of the transverse shear effect by the CPT.

4. Conclusions

In this article, a proposal of a new simpler formula of the Euler buckling stress of isotropic rectangular plates under axial compression loading in one and two orthogonal directions. The formula determined is similar to that of Bryan's one. The present formula is determined by applying the first shear order deformation plate theory proposed by Meksi *et al.* (2015), which taking into account the transverse shear effect in a uniform manner across the thickness of the plate, which necessitates the introduction of a shear correction factor k_s. Some numerical applications have been presented and compared with those obtained by the finite elements method. Finally, it can be concluded that the introduction of a corrective shear function makes it possible to obtain the results closer to those of the FEM.

This work can be considered as a basis for future works where high order theories (Yahia *et al.* 2015, Attia *et al.* 2015, Belkorissat *et al.* 2015, Mahi *et al.* 2015, Ahouel *et al.* 2016, Beldjelili *et al.* 2016, Kolahchi *et al.* 2017b, c,

Table 5 The Euler buckling load N_E of isotropic rectangular plate (*Cas*3−α=4.00, γ=1.0)

t/*b*	Mean element length	ANSYS Piscopo (2010) (A)	CPT Piscopo (2010) (B)	HSDT Piscopo (2010) (C)	FSDT Present (D)	$\frac{B-A}{A}*100$	$\frac{C-A}{A}*100$	$\frac{D-A}{A}*100$
---	m	KN/m	KN/m	KN/m	KN/m	%	%	%
0.01	0.050	199	198	198	198	0.00	0.00	0.00
	0.025	198						
	0.010	198						
	0.005	198						
0.02	0.050	1585	1583	1581	1581	0.51	0.38	0.38
	0.025	1580						
	0.010	1577						
	0.005	1575						
0.03	0.050	5337	5341	5327	5327	0.70	0.43	0.43
	0.025	5317						
	0.010	5309						
	0.005	5304						

Menasria *et al.* 2017, Hachemi *et al.* 2017, Bellifa *et al.* 2017a, b, Benadouda *et al.* 2017, Hajmohammad *et al.* 2018b, c, Hosseini and Kolahchi 2018, Karami *et al.* 2018d, Kaci *et al.* 2018, Attia *et al.* 2018, Bakhadda *et al.* 2018, Fourn *et al.* 2018, Bouadi *et al.* 2018, Belabed *et al.* 2018, Meksi *et al.* 2019, She *et al.* 2019) can be used or other types of materials will be considered. In addition, the current study gives a good foundation for extension to more general computational simulation for more complex geometrical configurations such as shells structures (Zine *et al.* 2018, Karami *et al.* 2018e) and very thick plates (Bousahla *et al.* 2014, Belabed *et al.* 2014, Hebali *et al.* 2014, Bennai *et al.* 2015, Meradjah *et al.* 2015, Chaht *et al.* 2015, Bourada *et al.* 2015, Hamidi *et al.* 2015, Bourada *et al.* 2015, Bennoun *et al.* 2016, Draiche *et al.* 2016, Bouafia *et al.* 2017, Karami *et al.* 2018f, g, Benahmed *et al.* 2017, Sekkal *et al.* 2017b, Benchohra *et al.* 2018, Younsi *et al.* 2018, Abualnour *et al.* 2018, Bouhadra *et al.* 2018, Mahmoudi *et al.* 2019, Zaoui *et al.* 2019).

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